STUDY OF MOTOR CYCLE ENGINE CHARACTERISTICS

BY

C. L. OTT

S. PIMSTEIN

ARMOUR INSTITUTE OF TECHNOLOGY

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A study of motor cycle
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A STUDY OF MOTOR CYCLE ENGINE CHARACTERISTICS

A THESIS

PRESENTED BY

C. L. OTT
S. PIMSTEIN

TO THE

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FOR THE DEGREE OF

BACHELOR OF SCIENCE IN MECHANICAL ENGINEERING

HAVING COMPLETED THE PRESCRIBED COURSE OF STUDY IN

MECHANICAL ENGINEERING

MAY 25, 1916

L. A. Morin
Dean of Cult. Studies

O. K.
F. W. Gebhardt

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PREFACE.

With the advent of the electric dynamometer, which may be run up to 3500 R.P.M., a very reliable and efficient means of testing high speed gas engines has been made possible. It is the purpose of this paper to give the procedure and results of tests made with this dynamometer on two well known makes of motorcycle engines.

CHAPTER 1.

METHOD USED IN TESTING.

SYNOPSIS:- A discussion of the electric dynamometer; engine stands; speed indicators; and torque measurement.

THE ELECTRIC DYNAMOMETER:

The electric dynamometer consists mainly of an electric generator whose field is balanced and free to move within small limits about its axis, which is supported on two roller bearings. The armature of the generator is coupled to the crankshaft of the engine to be tested. As the armature revolves in the magnetic field, it tends to draw the field about with it. The amount of this pull on the field can be measured by means of an extending arm fastened to the field frame, the arm resting on a scale. All of the power developed is thus used in cutting

lines of force, and, neglecting the windage of the armature and friction of the roller bearings, this power may be found by noting the speed of the armature and pull on the torque arm.

We have

$$\text{Horse power} = \frac{\text{Force x distance}}{\text{A constant}}$$

$$= \frac{6.28\ F\ r\ N}{33{,}000}$$

where

F = the pounds registered on scale.

r = radius of torque arm in feet.

N = revolutions per minute.

As may be anticipated, such a machine might be supplied with outside current and used to crank the engine for starting. Such was the case with the one used. A detailed description and enclosed blueprint (Fig. 1) give the wiring diagram.

The ammeter is connected in the armature circuit and the voltmeter across the supply

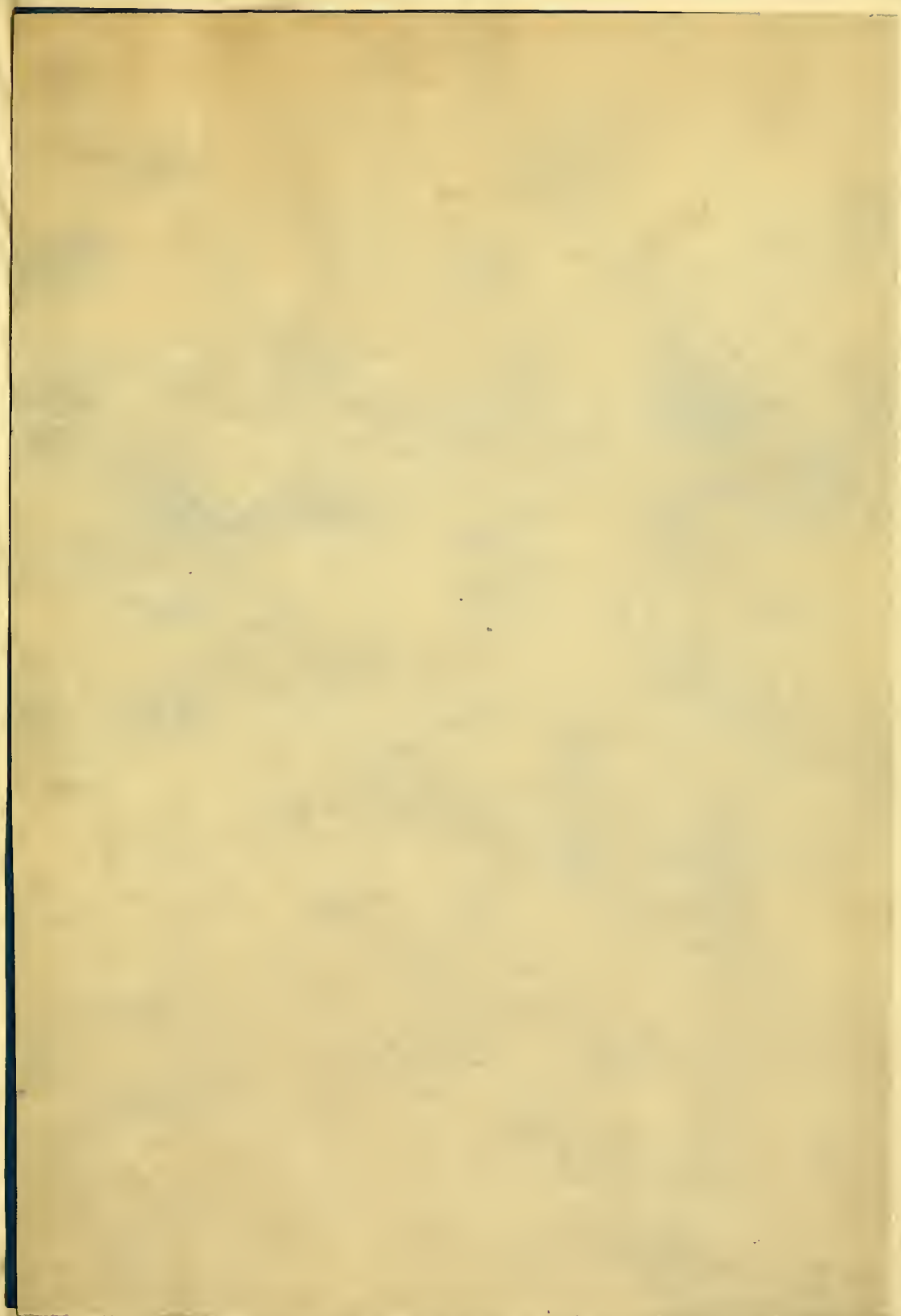
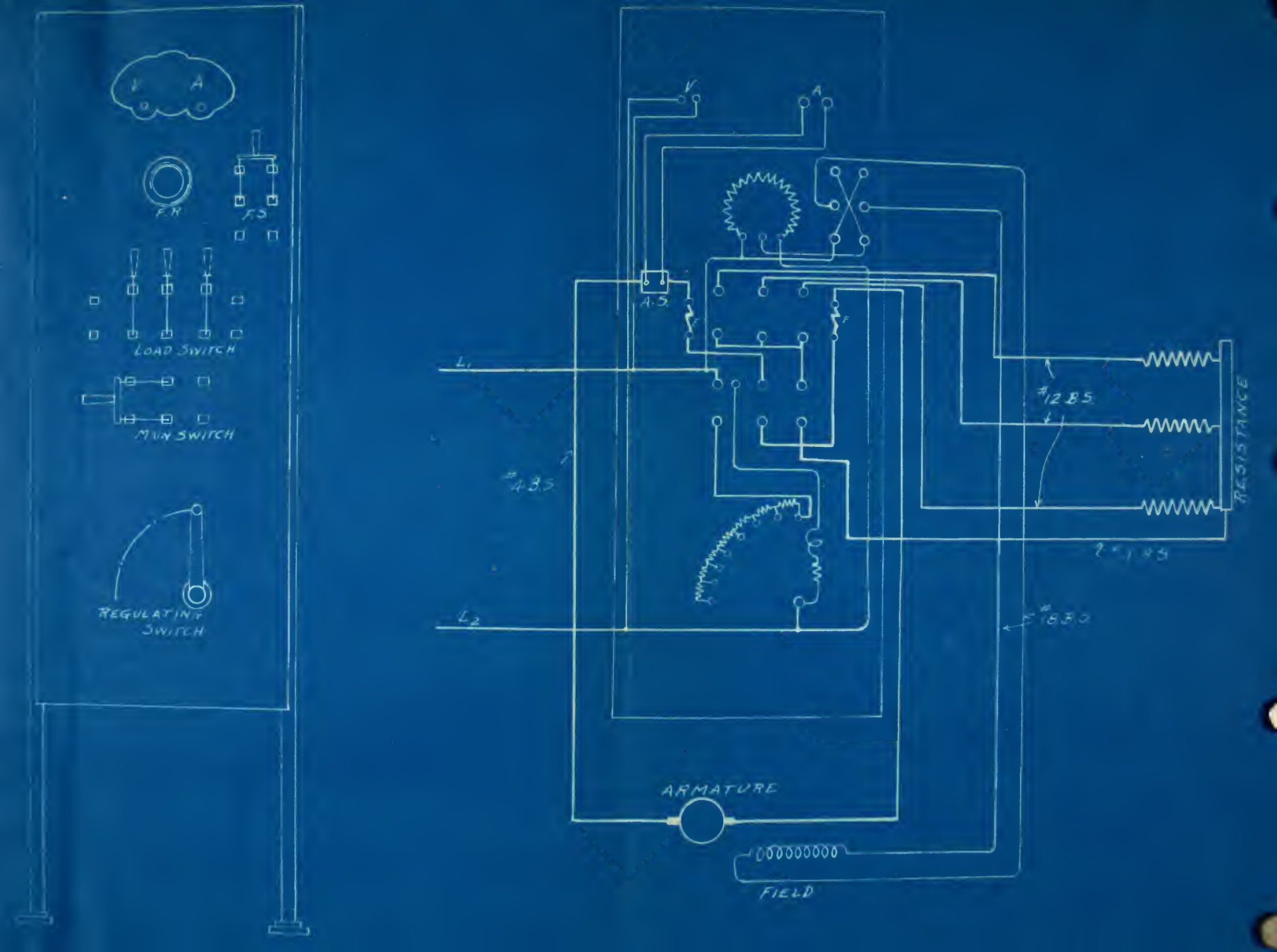


FIGURE 1.


V
A
F.R
F.S
LOAD SWITCH
MAIN SWITCH
REGULATING SWITCH
A.S.
F
L1
L2
#4 B.S.
#12 B.S.
#18 B.S.
RESISTANCE
ARMATURE
FIELD

circuit. The field rheostat is connected directly across the line, and the field of the dynamometer is connected to the middle point and one outside point of the field rheostat thru a reversing switch, the object of which is to change the rotation of the dynamometer as a motor, without having to change any connections. The three vertical switches connect the dynamometer to the load when it is running as a generator. The horizontal switch, when thrown to the left, connects the dynamometer to the line, so that it may be operated as a motor; and when thrown to the right, connects the dynamometer to the load resistance thru the three load switches.

Before operating the dynamometer, the direction of rotation of the engine should be ascertained, the torque arm and scale placed on the correct side for a brake horse power test, and the throttle and spark set to facilitate easy cranking and starting. All switches on the panel should be opened. Close in the field

switch on the correct side. Throw the main switch to the left. The dynamometer is then brought up to speed by means of the regulating switch. The engine should start now and its speed should be kept down until the dynamometer is connected up as a generator. To operate the dynamometer as a generator, throw the main switch to the right and adjust the load by means of the three load switches and field rheostat handle. The load switches put more coils in parallel with the armature, thus increasing the armature current, while the field rheostat may be used to put more resistance in the field, thus weakening it and making the load on the machine smaller. By the use of this field rheostat only, the speed of the engine with wide open throttle may be raised or lowered at will.

Instantaneous readings of speed and torque should be taken, and, since the torque arm is 31 1/2 inches long, the horse power may

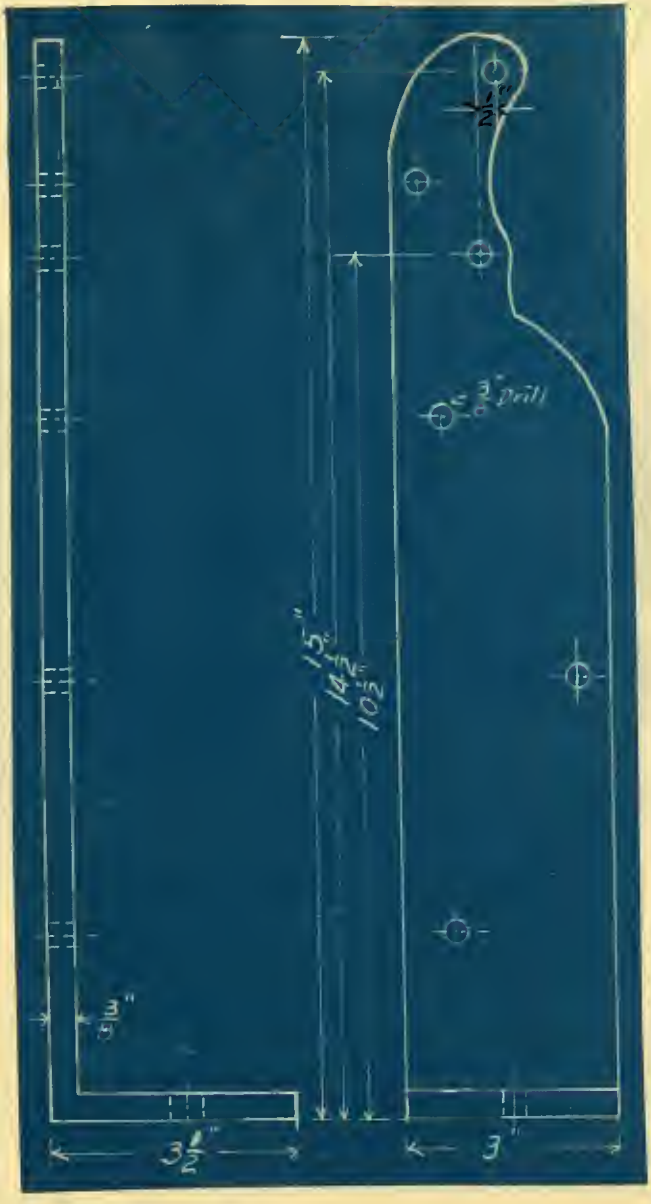


FIGURE 2.

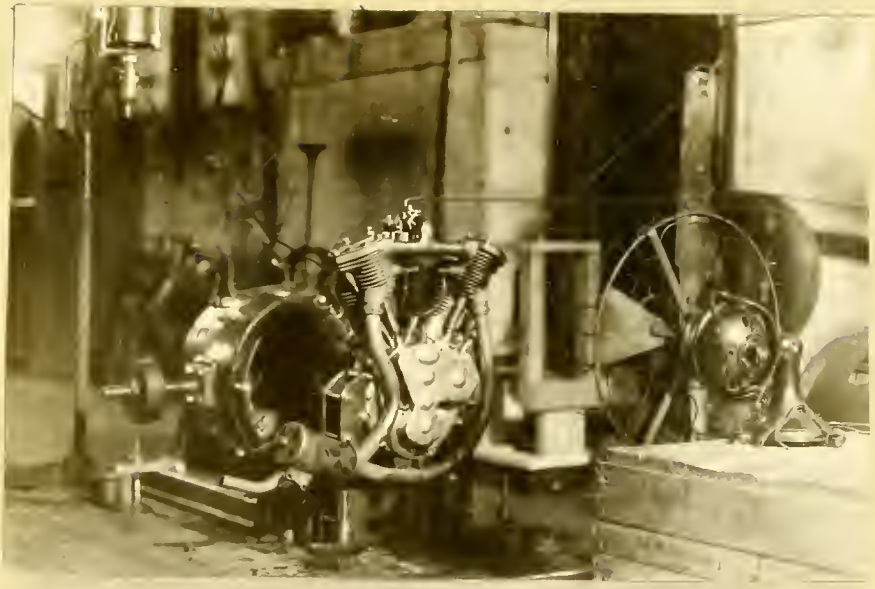


FIGURE 3.

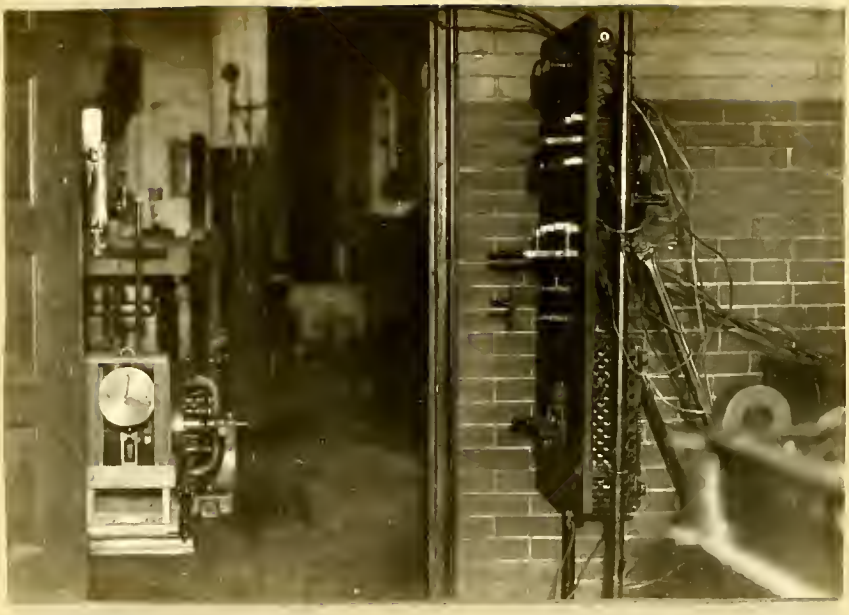


FIGURE 4.

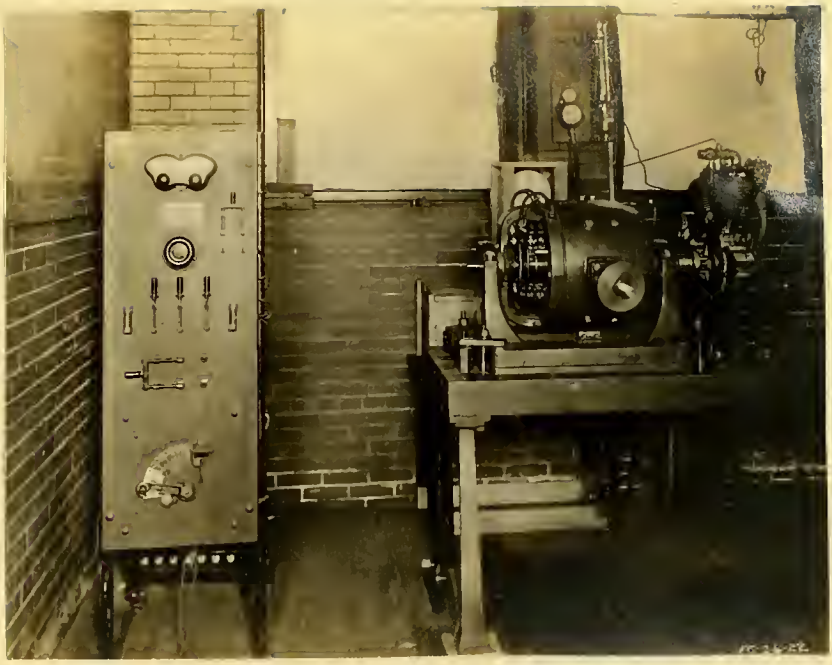


FIGURE 5.

be found from the formula

$$H.P. = \frac{6.28\ F\ r\ N}{33{,}000}$$

$$= \frac{6.28 \times 31.5 \times N}{33{,}000 \times 12}$$

$$= \frac{F\ N}{2{,}000} .$$

The friction horse power of the engine with wide open throttle may be found by measuring the torque and speed of the dynamometer when run as a motor. The torque arm in this case is on the opposite side from the one used for the brake horse power tests.

ENGINE STANDS AND COUPLINGS:

The engine must be supported alongside of the dynamometer and coupled to it. For the Thor Twin a special stand was made, a drawing of which is given (Fig. 2), and the complete stand is shown in the photograph of the Thor set up (Fig. 3). 3/8 in. bolts and 1/2 in. pipe were used to stiffen the stand. The coupling was made as shown in the print (Fig. 6), and its six projecting

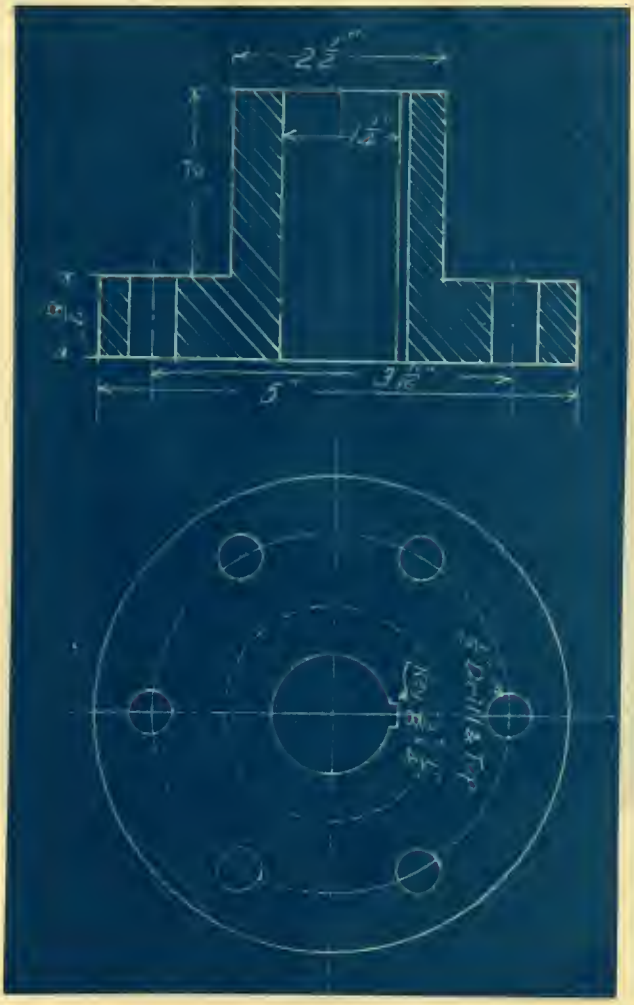
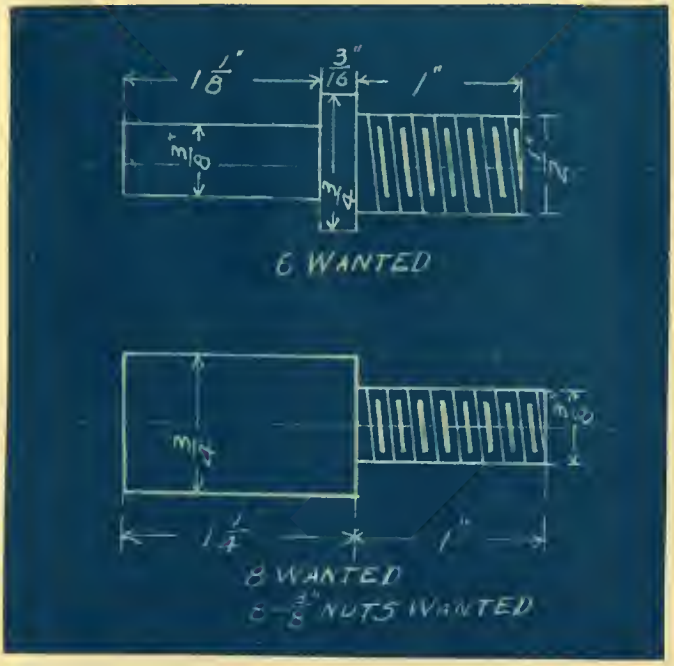


FIGURE 6.



7 a

7 b

FIGURE 7.

line (Fig. 7a) meshed in the eighteen tooth sprocket wheel of the engine. This sort of coupling was found to be too rigid, and, altho considerable clearance was allowed for the pins, trouble was experienced in shearing the sprocket wheel key.

A special frame was shipped with the Harley-Davidson engine, and this proved to be very satisfactory. A flexible four disc raw hide coupling was made for this engine(Fig. 8) and no trouble of shearing the crankshaft key was experienced. Figure 7b shows the pins used in this coupling.

SPEED INDICATORS:

An electric counter was used on the dynamometer at first. This was a continuous counter arranged so that it might be thrown in by pressing a button; and at the end of a half minute, the current being cut off by again pressing a button, the solenoid would let go; and the difference between the counter

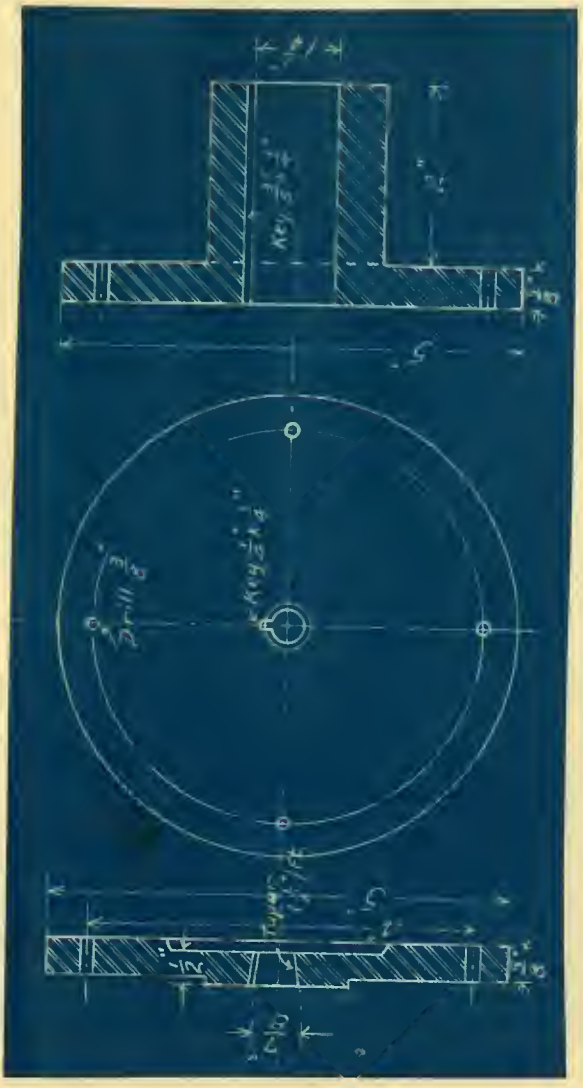
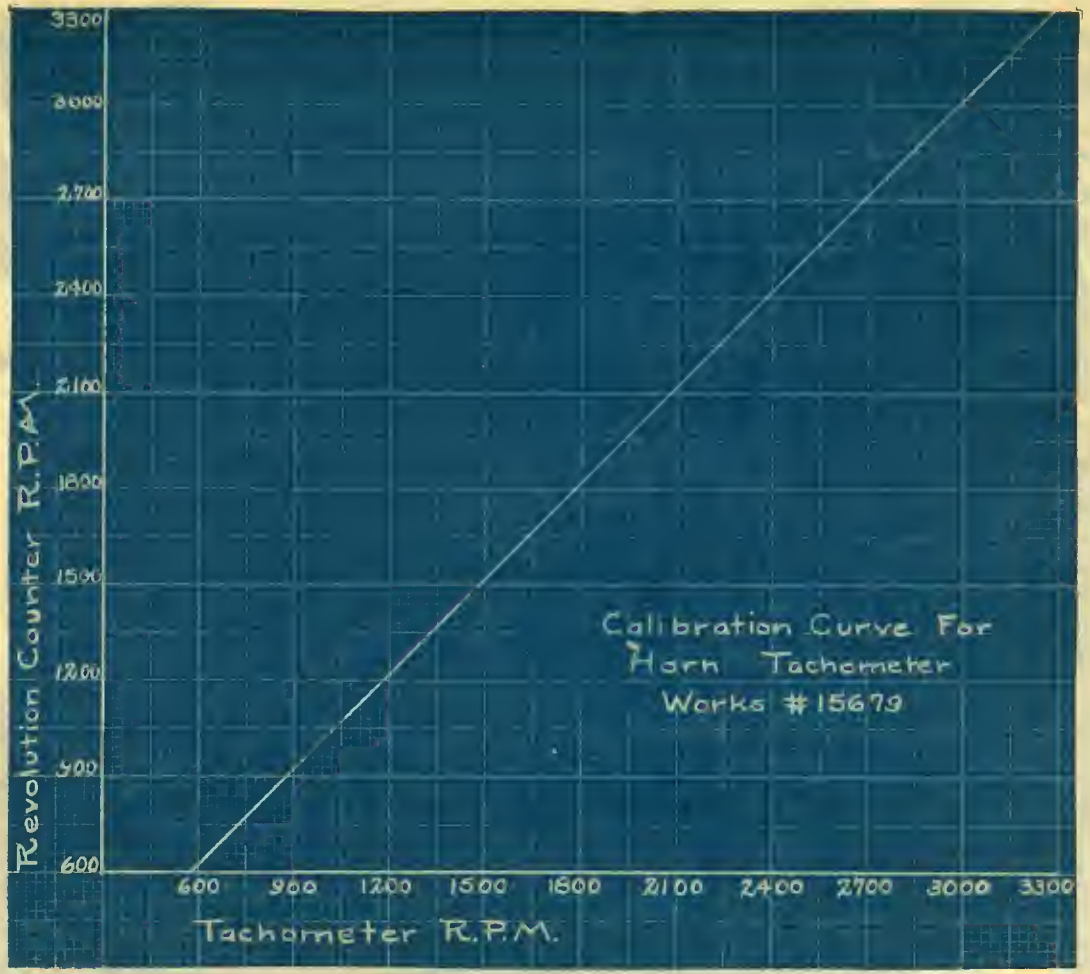


FIGURE 8.

number at the start and that at the end of the half minute would be the number of revolutions during that time. This would be alright for a test run; but in order to tell about at what speed the engine was running, a speedometer was connected by a belt and a suitable pulley to the dynamometer shaft, the speedometer indicating sixty miles per hour when the engine was running at three thousand revolutions per minute; fourty miles per hour at two thousand r.p.m., etc. It required some power to drive this speedometer and, as the clutching mechanism on the electric counter slipped at times, a calibrated tachometer was finally resorted to. A calibration curve of such a tachometer is given in Figure 9, and as the average error is less than one per cent, this method of taking speed was adopted.

TORQUE MEASUREMENT:

A calibrated spring balance was used in taking torque readings. This balance read


Revolution Counter R.P.M.
3300
3000
2700
2400
2100
1800
1500
1200
900
600
600
900
1200
1500
1800
2100
2400
2700
3000
3300
Tachometer R.P.M.
Calibration Curve For
Horn Tachometer
Works #15679

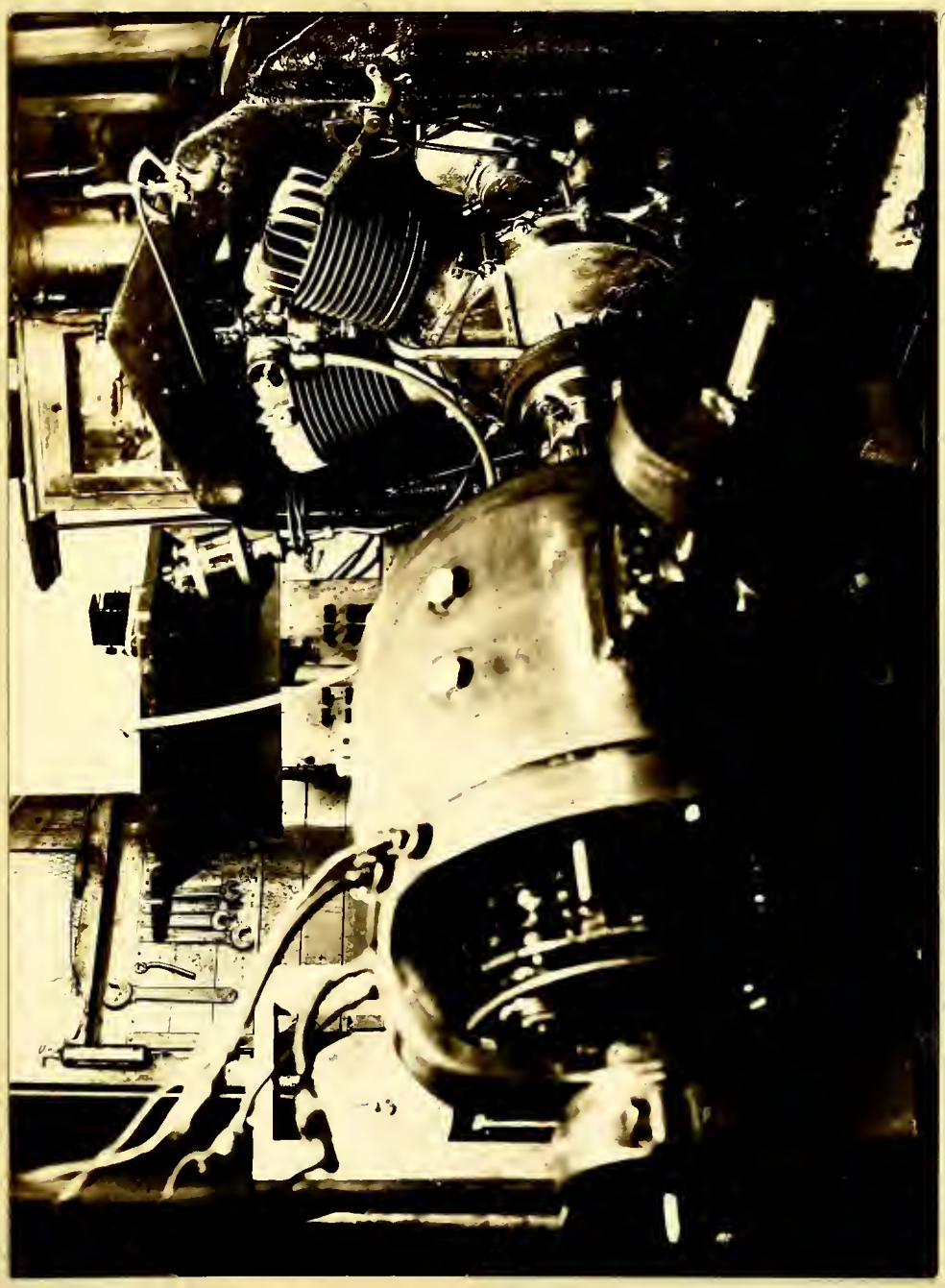


FIGURE 1[illegible].

to tenths of a pound, thus making an error of not more than one per cent on those readings. Great care was taken to have the incoming leads to the dynamometer flexible and the dynamometer in balance when running free, the balance weight being used for adjustment. The effective torque arm was 31 1/2 inches long.

GASOLINE SUPPLY AND CONTROLS:

Ordinary gasoline, 55° Baume, was used at all times and for lubrication Texaco special oil. An ordinary gas tank was employed, the head on the carburetor being about two feet. Spark and throttle controls were made as illustrated in the photographs of the set ups (Figs. 3, 5, and 10).

CHAPTER 2.

TEST OF THOR TWIN.

SYNOPSIS:- The maximum power curve, the friction horse power, and the valve lift diagrams were obtained.

ENGINE:

The engine tested was a stock twin, 716 U 676, with cylinders set at 50°, the magneto being forward of the front cylinder.

Piston displacement------59.7 cu.ins.
Bore---------- 3.25 ins.
Stroke---------- 3.60 ins.
Inlet valves, diameter--- 1.375 ins.
Exhaust valves, dia.----- 1.50 ins.
One inch Schebler carburetor.
Bosch magneto.
Air cooled.
Splash system of lubrication.
A.L.A.M. rating at 1370 R.P.M.-8.44 hp.

TEST:

The engine was mounted on a special stand, coupled to the dynamometer (Figs. 3-5), and run with an eighteen inch electric fan playing on the cylinders. Gasoline, 55° Baume,

was used for fuel and Texaco Special Oil for lubrication. The engine was allowed to exhaust thru an open muffler into the room. Maximum power was aimed at for all speeds and the carburetor and spark were set to give the desired result. The maximum spark advance was 70°. Sufficient lubrication was provided for at all times, as this is quite essential with an air cooled engine having only stock clearance.

Instantaneous torque and speed readings were taken, allowing about a minute each time for conditions to become steady. For complete data of these tests see Table No. 1 and Figure No. 11.

A friction horse power test was made with wide open throttle and with the ignition on. Sufficient lubrication was provided for in this run also. The results of this test are given in Table No. 2 and Figure No. 12.

A maximum horse power curve was drawn as in Figure No. 11 and combining this with

TABLE I.

LOG OF TEST DATA ON THOR TWIN #16 U 676 .

SPEED R.P.M.	TORQUE # AT 31 1/2"	B.H.P.
1250	10.5	6.56
1300	11.0	7.15
1400	12.2	8.54
1500	10.0	7.50
1550	9.5	7.36
1694	10.5	8.73
1698	7.1	6.02
1750	11.1	9.71
1760	7.6	6.69
1892	9.4	8.90
1950	8.5	8.28
2000	8.5	8.50
2022	8.0	8.09
2100	7.0	7.35
2125	7.3	7.76
2150	8.0	8.60
2200	9.0	9.90
2250	7.6	8.55
2250	8.0	9.00
2300	9.4	10.81
2300	8.2	9.45
2338	6.8	7.95
2348	7.5	8.80
2352	6.9	8.10
2450	8.8	10.79
2600	7.2	9.35
2600	8.5	11.05

TABLE 1.

LOG OF TEST DATA ON THOR TWIN #16 U 676, CONT.

SPEED R.P.M.	TORQUE # AT 31 1/2"	B.H.P.
2650	6.20	8.20
2650	6.25	8.28
2650	7.50	9.95
2650	8.10	10.73
2650	7.00	9.27
2700	6.00	8.10
2700	7.00	9.45
2850	6.20	8.83
2870	6.50	9.32
2900	6.30	9.13
3000	5.00	7.50
3000	5.50	8.25
3050	5.16	7.80
3050	6.40	9.76
3100	4.50	6.97
3200	5.80	9.28

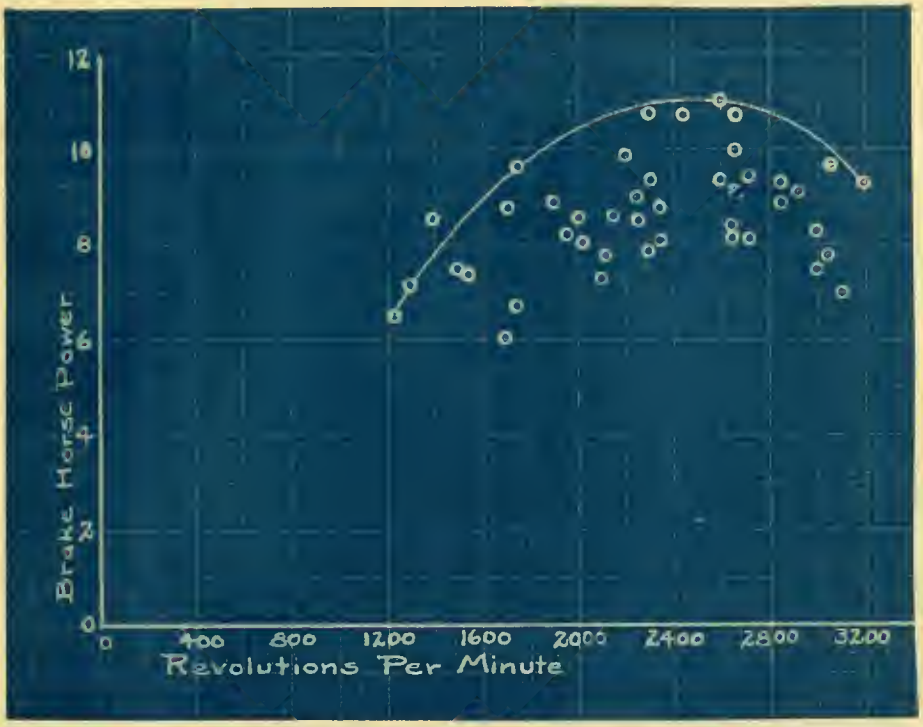

12
10
8
6
4
2
0
Brake Horse Power
0
400
800
1200
1600
2000
2400
2800
3200
Revolutions Per Minute

FIGURE 11.

TABLE 1.

FRICTION HORSE POWER TEST OF THOR TWIN.

SPEED R.P.M.	TORQUE # AT 31 1/2'	H.P.
1200	2.7	1.62
1400	2.8	1.96
1670	3.1	2.515
1970	3.4	3.35
2280	3.8	4.33
2680	4.5	6.04

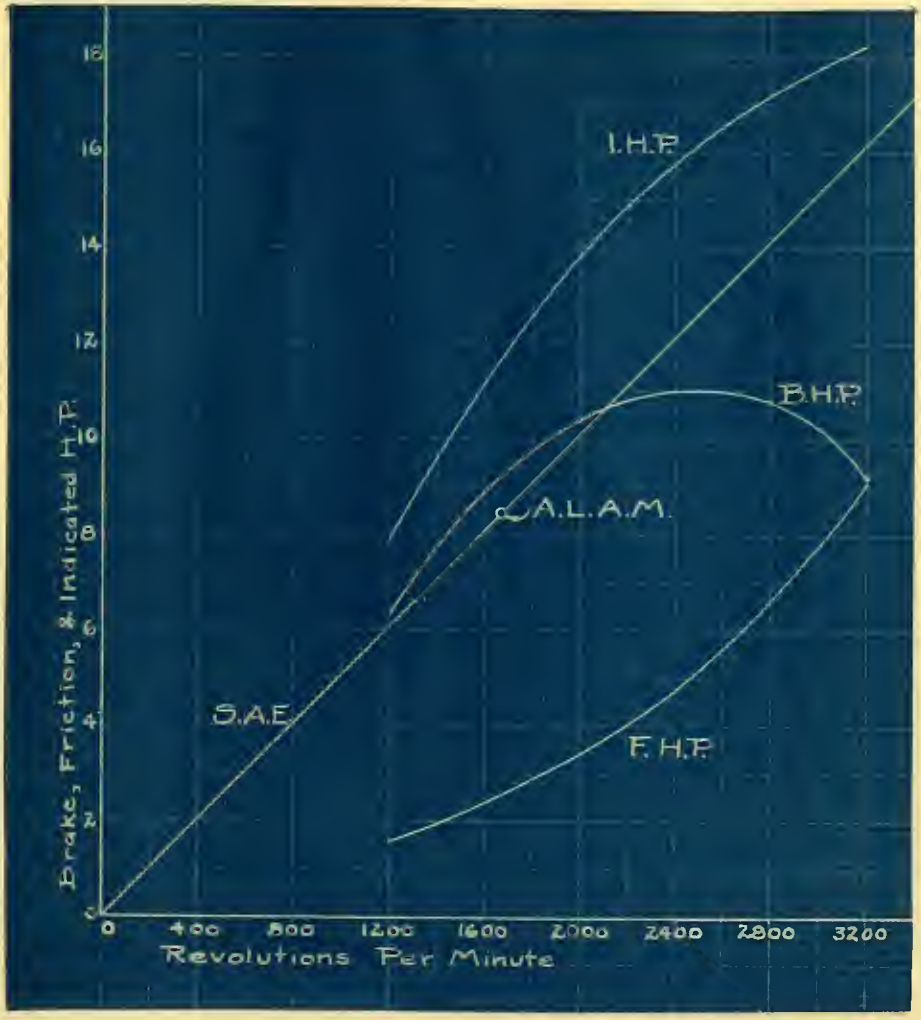

Brake, Friction, & Indicated H.P.
18
16
14
12
10
8
6
4
2
0
I.H.P.
B.H.P.
A.L.A.M.
S.A.E.
F.H.P.
0
400
800
1200
1600
2000
2400
2800
3200
Revolutions Per Minute

FIGURE 12.

the friction horse power, the cylinder horse power curve was constructed as shown in Figure 12. The torque and mean effective pressure net curves (Fig. 13) were calculated from data taken from the maximum power curve (See Table 3).

SAMPLE CALCULATIONS:

A.L.A.M. rating at 1000 ft. per min. piston speed is given by the formula

$$\text{Horse power} = \frac{d^2 n}{2.5}$$

where d = the bore in inches.

n = the number of cylinders.

For this engine, 1000 ft. per min. piston speed $= \frac{12 \times 1000}{2 \times 3.6} = 1670$ r.p.m., and

$$\text{H.P.} = \frac{(3.25)^2 \times 2}{2.5} = 8.44.$$

The brake horse power $= \frac{F\,N}{2,000}$.

At 2450 r.p.m. F = 8.8 pounds (Table 1).

Therefore

$$\text{H.P.} = \frac{8.8 \times 2450}{2,000} = 10.78.$$

TABLE 3.

DATA TAKEN FROM MAXIMUM POWER CURVE FOR THE TWIN 16 U 676.

SPEED R.P.M.	B.H.P.	TORQUE AT 1 FT.	M.E.P. NET PER SQ.IN.
1200	6.4	28.0	70.8
1400	7.7	28.9	73.0
1600	8.9	29.2	73.8
1800	9.75	28.4	71.8
2000	10.4	27.3	69.0
2200	10.8	25.8	65.1
2400	11.0	24.1	60.9
2600	11.0	22.2	56.1
2800	10.8	20.3	51.2
3000	10.3	18.0	45.5
3200	9.2	15.1	38.1

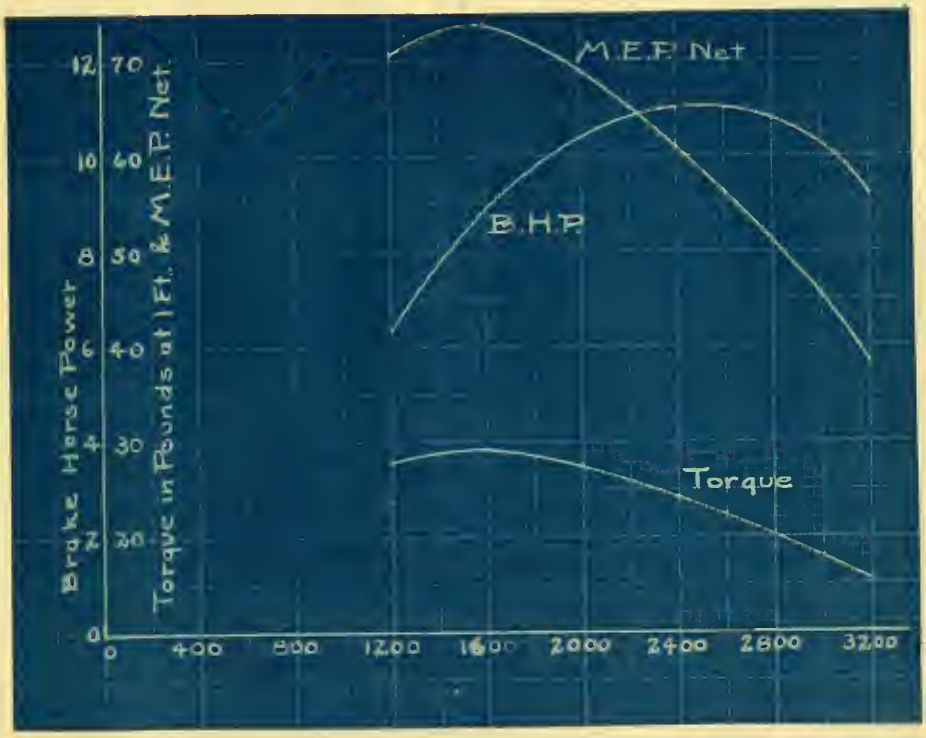

M.E.P. Net
B.H.P
Torque
Brake Horse Power
Torque in Pounds at 1 Ft. & M.E.P. Net
0
2
4
6
8
10
12
20
30
40
50
60
70
0
400
800
1200
1600
2000
2400
2800
3200

FIGURE 13.

The mean effective pressure net may be found from the relation

$$B.h.p. = I.h.p. \times Mechanical\ eff.$$

$$= \frac{P\ L\ A\ N \times Mech.\ eff.}{33,000}$$

where

P x Mech. eff. = the M.e.p. Net.

L = the stroke in feet.

A = the area of the piston in sq.ins.

N = the number of working strokes per minute = the r.p.m. in this case.

Supplying the values for L and A and solving for the M.e.p. Net, we have

$$M.e.p.\ Net = \frac{33,000 \times 12 \times B.h.p.}{3.6 \times .7854(3.25)^2 \times R.p.m.}$$

$$= 13,280\ \frac{B.h.p.}{R.p.m.}\ .$$

An expression for the torque in pounds at one foot may be derived from the relation

$$B.h.p. = \frac{6.28\ F\ r\ N}{33,000}$$

Solving

$$Torque = F\ r = 5250\ \frac{B.h.p.}{R.p.m.}\ .$$

VALVE LIFT DIAGRAMS:

The dead center was found with a trammel, the removal of a plug in the cylinder head allowing one to reach the top of the piston with a rod. A 2 27/32 in. circle on the sprocket wheel was laid out in 15° intervals, and the events and the amount of valve lift were determined. For the complete data see Table No. 4 and Figure No. 14.

The port opening was calculated from the formula (Heldt's)

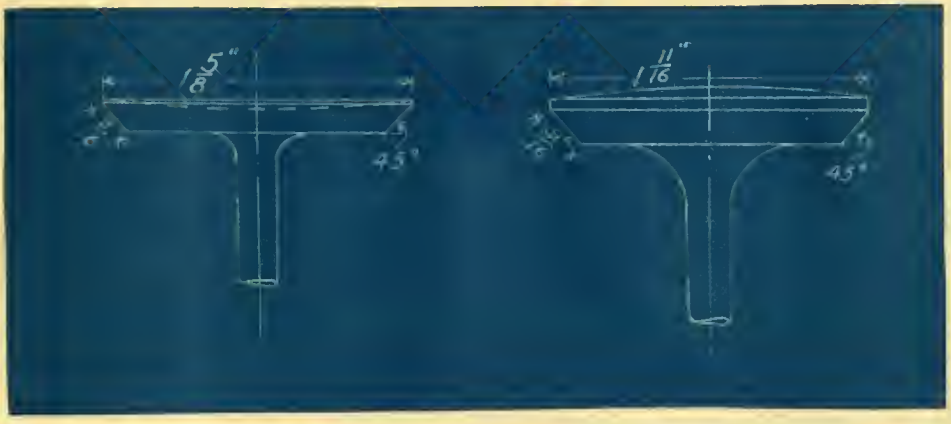
$$A = 3.1416(.707\ dh + .353\ h^2)$$

where

A = the area of the opening in sq. ins.

d = the diameter of the valve in the clear.

h = the valve lift in inches.



INLET EXHAUST

VALVES IN TIGER TWIN.

FIGURE 13a.

TABLE 4.

VALVE LIFT DATA FOR [illegible] 16 U 676.

NORMAL SPEED.

INLET VALVE.

DEGREES PAST D.C.	LIFT--INS.	PORT OPENING-SQ.INS.
16°	0	0
30	.04	.1236
45	.10	.317
60	.14	.450
75	.16	.515
90	.16	.515
105	.16	.515
120	.15	.487
135	.15	.487
150	.15	.487
165	.13	.410
180	.10	.317
195	.06	.1855
210	.03	.093
225	0	0

TABLE 4, CONTINUED.

FORWARD CYLINDER.

EXHAUST VALVE.

DEGREES PAST DEAD CENTER	LIFT INS.	PORT OPENING SQ.INS.
119°	0	0
135	.06	.203
150	.12	.415
165	.17	.598
180	.22	.780
195	.25	.900
210	"	"
225	"	"
240	"	"
255	"	"
270	"	"
285	.245	.883
300	.22	.780
315	.17	.598
330	.13	.430
345	.08	.273
360	.03	.101
19	0	0

TABLE 4, [illegible].

[illegible] CYLINDER.

INLET VALVE		EXHAUST VALVE	
PORT OPENING SQ. INS.	% OF PISTON TRAVEL	PORT OPENING SQ. INS.	% OF PISTON TRAVEL
0	2%	0	81%
.1238	8	.203	88
.317	18	.415	95
.450	39	.598	98.5
.515	44	.780	100
.515	58	.900	1.5
.515	70	.900	
.487	81	.900	
.487	88	.900	
.487	95	.900	
.410	98.5	.900	42
.317	100	.893	56
.1855	1.5	.780	69
.093	5	.598	81
0	10	.450	92
		.273	98
		.101	100
		0	1

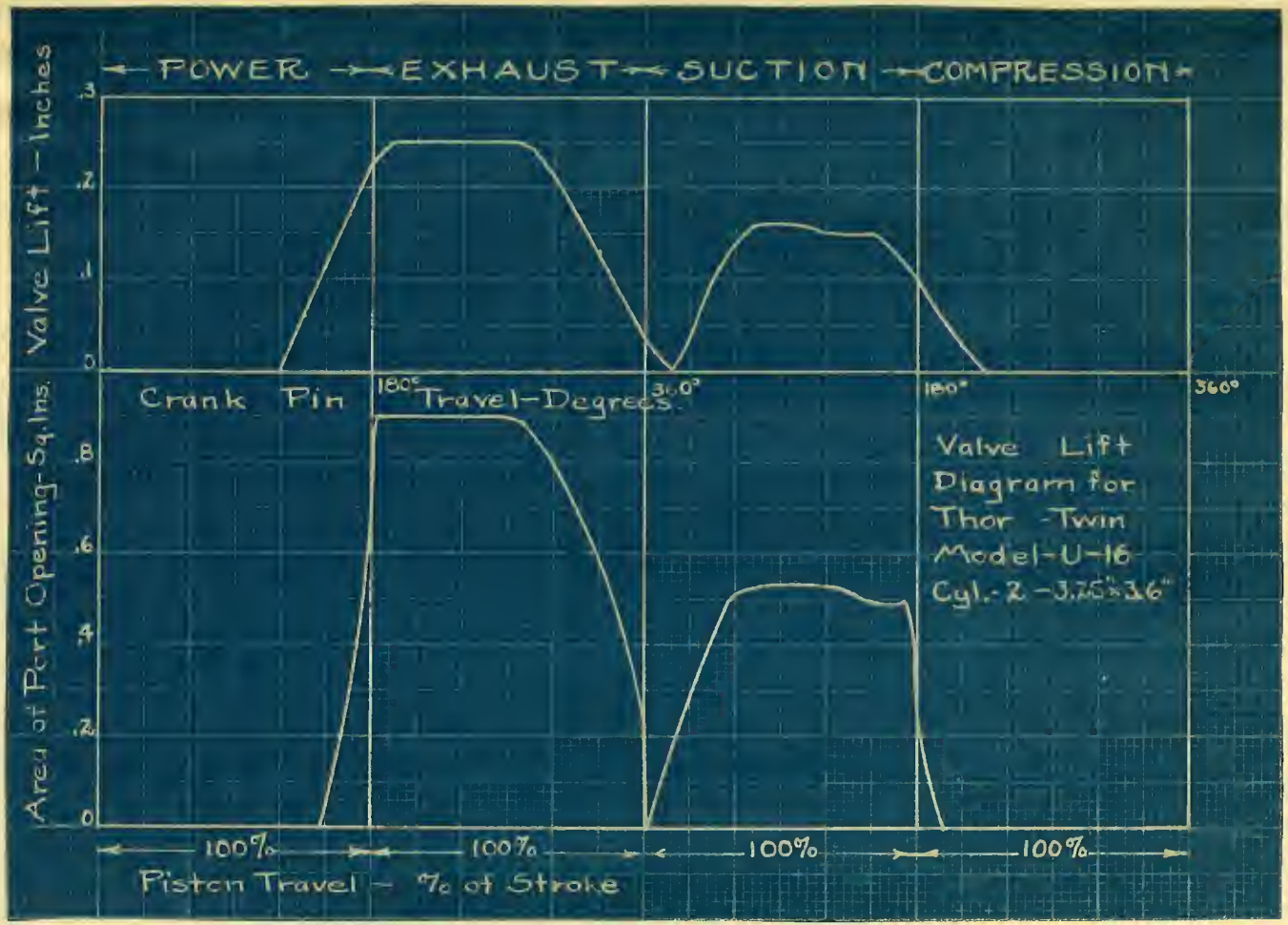

← POWER → ← EXHAUST ← SUCTION → COMPRESSION →
Valve Lift – Inches
.4
.2
0
Crank Pin
180° Travel – Degrees 360°
180°
360°
Valve Lift Diagram for Thor – Twin Model – U – 16 Cyl. – 2 – 3.25" × 3.6"
Area of Port Opening – Sq. Ins.
.8
.6
.4
.2
0
100%
100%
100%
100%
Piston Travel – % of Stroke

FIGURE 14.

TABLE 4, CONTINUED.

CAM OUTLINES.

INLET VALVE		EXHAUST VALVE	
DEGREES PAST DEAD CENTER	LIFT INS.	DEGREES PAST DEAD CENTER	LIFT INS.
15°	0	180°	0
30	.05	135	.05
45	.10	150	.11
60	.14	165	.16
75	.15	180	.19
90	.16	195	.24
105	.16	210	.25
120	.16	225	.25
135	.15	240	.25
150	.15	255	.25
165	.15	285	.25
180	.14	300	.22
195	.08	315	.17
210	.04	330	.12
220	0	345	.07
		360	.03
		15	0

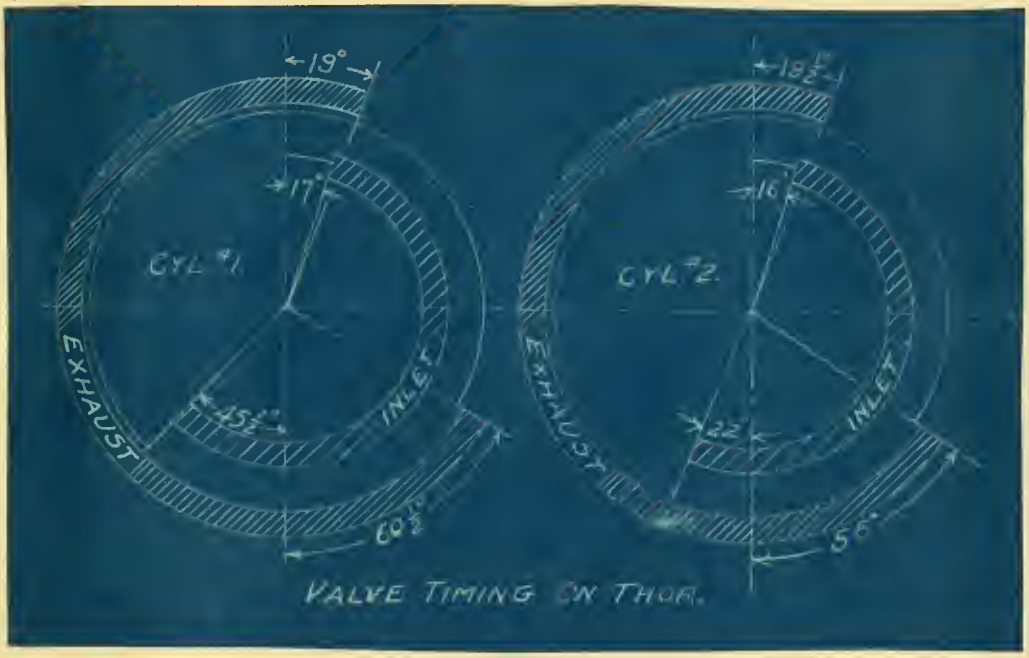

19°
17°
CYL. #1.
EXHAUST
INLET
45½°
60½°
19½°
16°
CYL. #2.
EXHAUST
INLET
22°
56°
VALVE TIMING ON THOR.

FIGURE 14a.

CHAPTER 3.

TEST OF HARLEY-DAVIDSON TWIN.

SYNOPSIS:- The maximum power curve, the friction horse power curve, and the valve lift diagrams were obtained.

ENGINE:

The engine tested was a stock twin, racing clearance, [illegible] 522 K, with cylinders set at 45°, the magneto being behind the back cylinder.

Piston displacement------60.34 cu.ins.
Bore---------- 3.3115 ins.
Stroke---------- 3.55 ins.
Inlet valves, diameter--- 1.5038 ins.
Exhaust valves, dia.----- 1.438 ins.
One inch Schebler carburetor.
Bosch magneto.
Air cooled.
Splash system of lubrication.
A.L.A.M. rating at 1715 R.P.M.-8.78 H.P.

TEST:

The engine, mounted in a special stand supplied by the Harley-Davidson Motor Company, was coupled to the dynamometer (Figs. 10

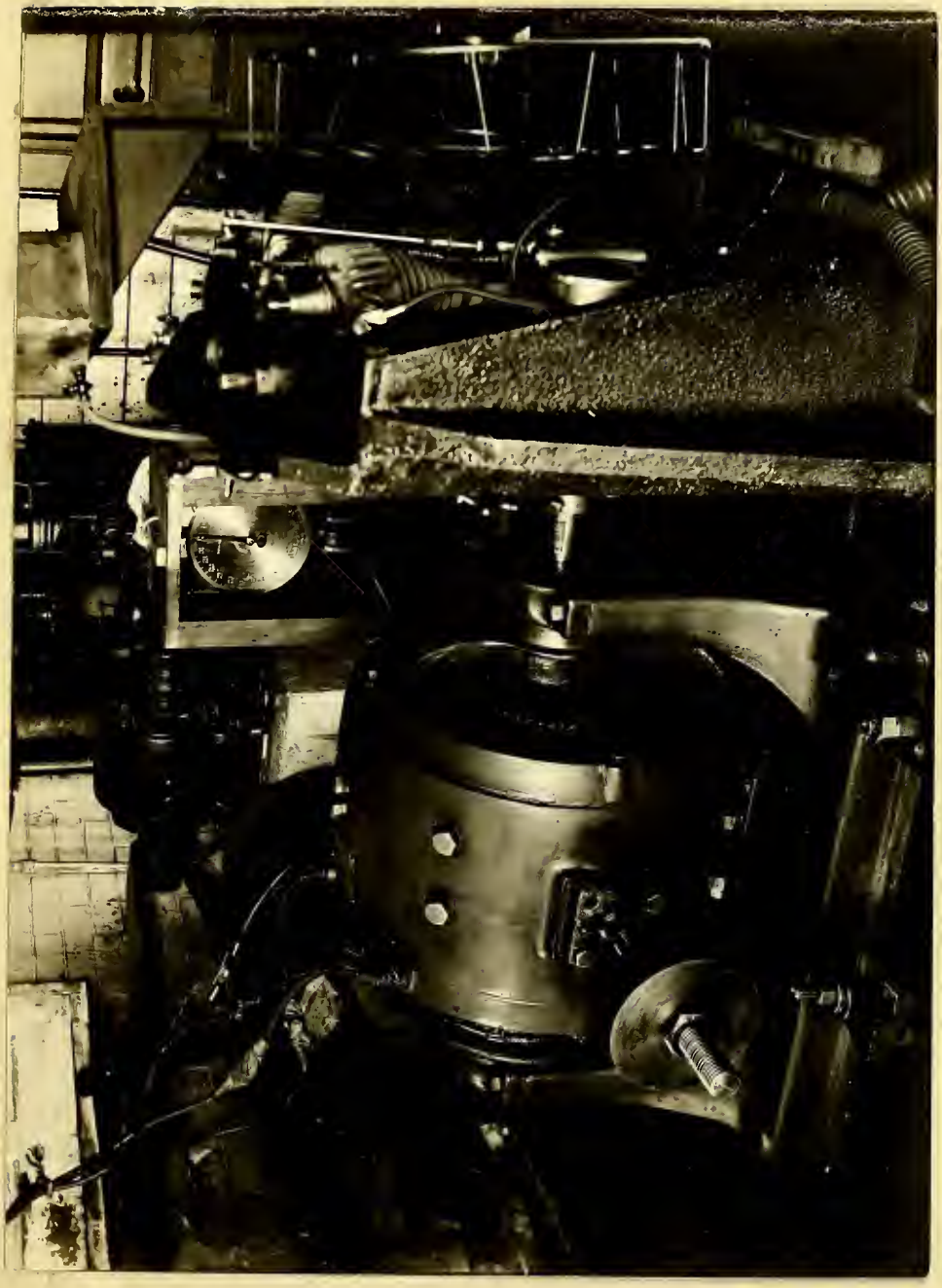


FIGURE 15.

and 15), and run with a current of air from an eighteen inch fan blowing on the cylinders. Gasoline, 55° Baume, was used for feul and Texaco Special oil for lubrication. The engine exhaust was piped to the outside air thru a three inch exhaust pipe. Maximum power was aimed for at all speeds, and the carburetor and magneto were set to produce this result. The maximum spark advance was 52°. Plenty of oil was used at all times.

Instantaneous torque and speed readings were taken, allowing about a minute each time for conditions to become steady. For complete data of these tests see Table No. 5 and Figure No. 16.

A friction horse power test was made with wide open throttle and with the ignition on. Sufficient lubrication was provided for in this run also. The results of this test are given in Table No. 6 and Figure No. 17.

A maximum horse power curve was drawn as in Figure No. 16, and combining this with

TABLE 4.

LOG OF TEST OF [illegible] ENGINE.

SPEED R.P.M.	TORQUE @ AT 51 1/2"	B.H.P.
1400	10.9	7.63
1400	10.4	7.28
1500	11.9	8.92
1750	10.9	8.99
1800	9.9	8.91
1800	12.4	11.17
1900	10.0	9.50
2100	10.2	10.70
2300	6.5	7.47
2350	10.4	12.99
2400	9.4	11.30
2500	9.3	11.63
2700	10.0	13.50
2700	9.8	13.38
2700	7.5	10.13
2800	7.5	10.50
2750	8.5	11.68
2800	9.7	13.60
2850	8.0	11.40
3000	8.8	10.20
3200	7.6	12.15
3200	6.9	11.02
3200	7.2	11.51
3250	8.5	13.72
3250	8.5	10.57
3300	6.6	10.89
3350	6.2	10.40
3350	7.4	12.40
3300	7.7	12.70
3375	6.0	10.10
3400	6.1	10.35
3500	6.8	11.90
3550	5.8	11.20

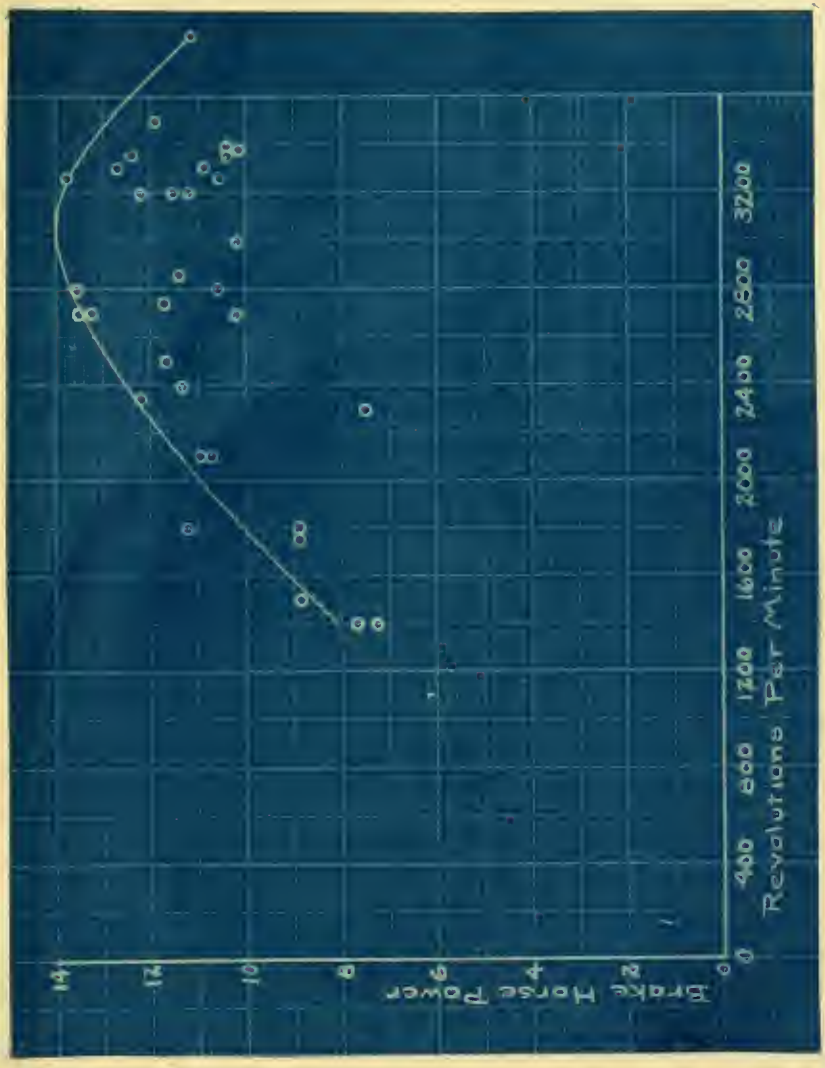

Brake Horse Power
0
2
4
6
8
10
12
14
Revolutions Per Minute
0
400
800
1200
1600
2000
2400
2800
3200

FIGURE 16.

TABLE 6.

FRICTION HORSE POWER TEST OF

HARLEY-DAVIDSON TWIN.

SPEED R.P.M.	TORQUE # AT 31 1/2"	F.H.P.
1600	3.0	2.40
1850	3.2	2.96
2100	3.3	3.46
2350	3.5	3.94
2600	3.4	4.42
3000	3.7	5.55

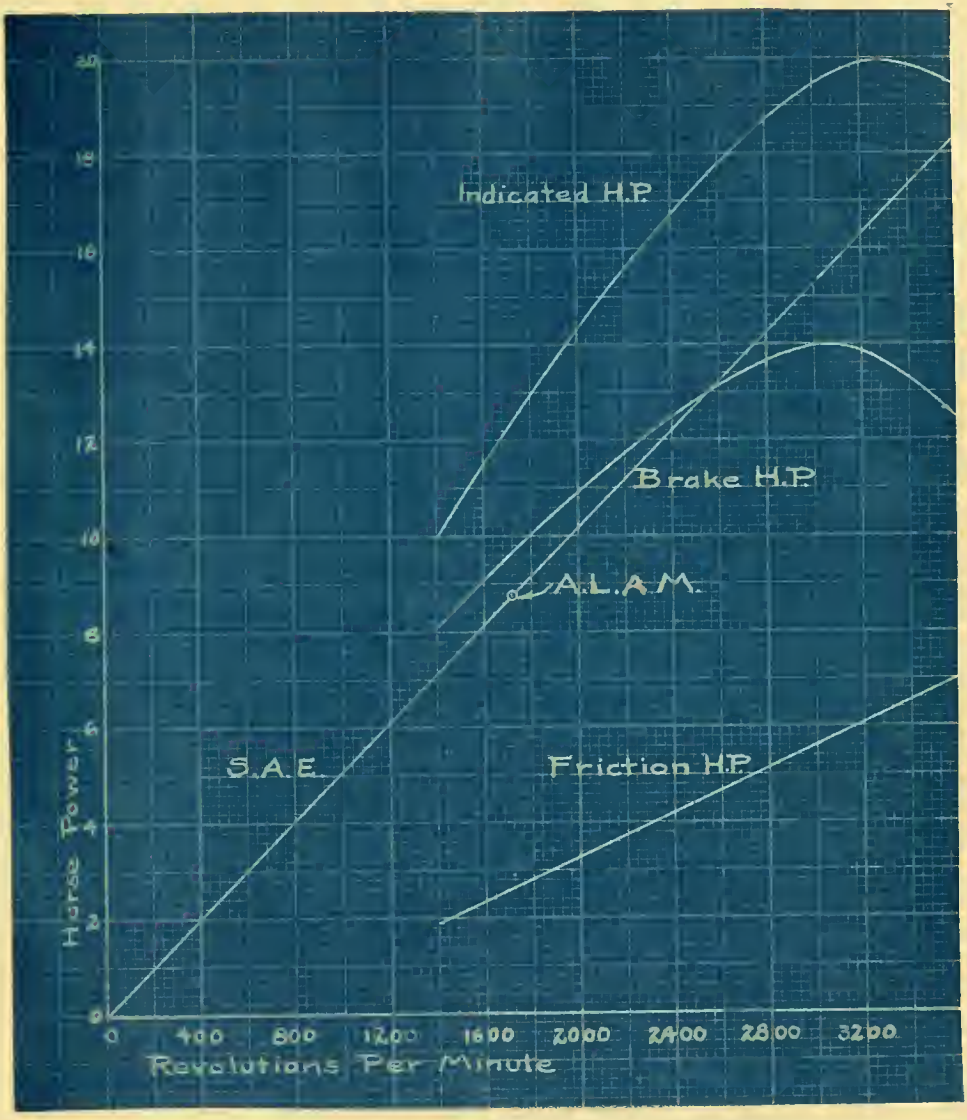

Indicated H.P.
Brake H.P.
A.L.A.M.
S.A.E.
Friction H.P.
Horse Power
0
2
4
6
8
10
12
14
16
18
20
0
400
800
1200
1600
2000
2400
2800
3200
Revolutions Per Minute

FIGURE 17.

the friction horse power, the cylinder horse power curve was constructed as shown in Figure 17. The torque and mean effective pressure net curves (Fig. 18) were calculated from data taken from the maximum power curve (See Table 7).

SAMPLE CALCULATIONS:

For the complete formulas and derivations of formulas used in the following calculations see Chapters 1 and 2.

A.L.A.M. rating at 1000 ft. per min. piston speed (1715 r.p.m.)

$$H.p. = \frac{d^2 n}{2.5}$$

$$= \frac{(3.313)^2 \times 2}{2.5}$$

$$= 8.78 \quad .$$

$$\text{Brake horse power} = \frac{F\ N}{2,000} \quad .$$

At 2700 r.p.m. F = 10 pounds.

and $$B.h.p. = \frac{10 \times 2700}{2,000} = 13.5 \quad .$$

TABLE 7.

B.H.P. TABLE FOR [illegible] FOR [illegible] FOR

HARLEY-DAVIDSON TWIN.

SPEED R.P.M.	B.H.P.	TORQUE # AT 1 FT.	M.E.P. LBT # PER SQ.IN.
1400	8.1	30.4	75.8
1600	9.1	29.8	74.5
2000	10.9	28.6	71.5
2400	12.5	27.4	68.4
2800	13.65	25.6	64.0
3000	14.00	24.5	61.4
3200	13.9	22.8	57.1
3400	13.5	20.5	51.3
3600	12.4	18.1	45.3

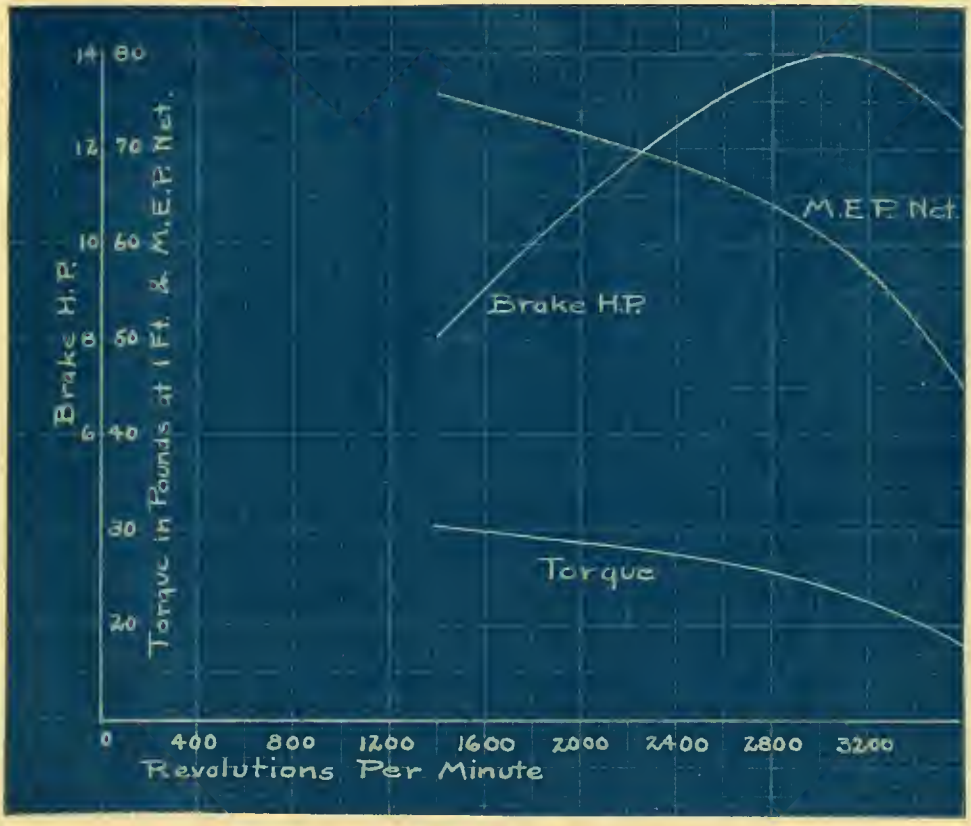

Brake H.P.
Torque in Pounds at 1 Ft. & M.E.P. Net.
14
12
10
8
6
80
70
60
50
40
30
20
M.E.P. Net
Brake H.P.
Torque
0
400
800
1200
1600
2000
2400
2800
3200
Revolutions Per Minute

FIGURE 18.

The mean effective pressure was found by applying the values in the formula as follows

$$\text{I.e.p. Net} = \frac{33,000 \times 13 \text{ B.h.p.}}{3.5 \times .7854(3.513)^2 \text{ R.p.m.}}$$

$$= 13,120 \frac{\text{B.h.p.}}{\text{R.p.m.}} .$$

The torque was calculated as before from the formula

$$\text{Torque in } \# \text{ at 1 ft.} = 5250 \frac{\text{B.h.p.}}{\text{R.p.m.}} .$$

VALVE LIFT DIAGRAMS:

The dead center was found with a trammel, the removal of a plug in the cylinder head allowing one to reach the top of the piston with a rod. The cylindrical face of the five inch coupling was laid out in 20° intervals, and the events and valve lifts were taken. For complete data see Tables 8 and 9 and Figures 19 and 20.

INLET EXHAUST

VALVES IN HARLEY-DAVIDSON TWIN.

FIGURE 18a.

TABLE 9

VALVE LIFT D.A. FOR TWIN-HUDSON TWIN.

FORWARD CYLINDER, INLET VALVE

CRANK PIN DEG. PAST D.C.	LIFT INS.	PORT OPEN-ING-SQ.INS.	% OF PISTON TRAVEL
- 13°	0"	0	98 %
0	.03	.1057	0
+ 20	.12	.454	6
40	.225	.842	15
60	"	"	31
80	"	"	53
100	"	"	66
120	"	"	81
140	.195	.723	95
160	.12	.454	97
180	.075	.2775	100
200	.015	.0525	2
208	0	0	5

FORWARD CYLINDER, EXHAUST VALVE

128	0	0	85
140	.01	.033	91
160	.02	.066	97.5
180	.12	.412	100
200	.21	.747	7
220	.24	.855	14
240	.25	.895	30
260	.22	.784	57
280	.20	.703	52
300	.14	.482	69
320	.08	.329	87
340	.04	.135	96
360	0	0	100

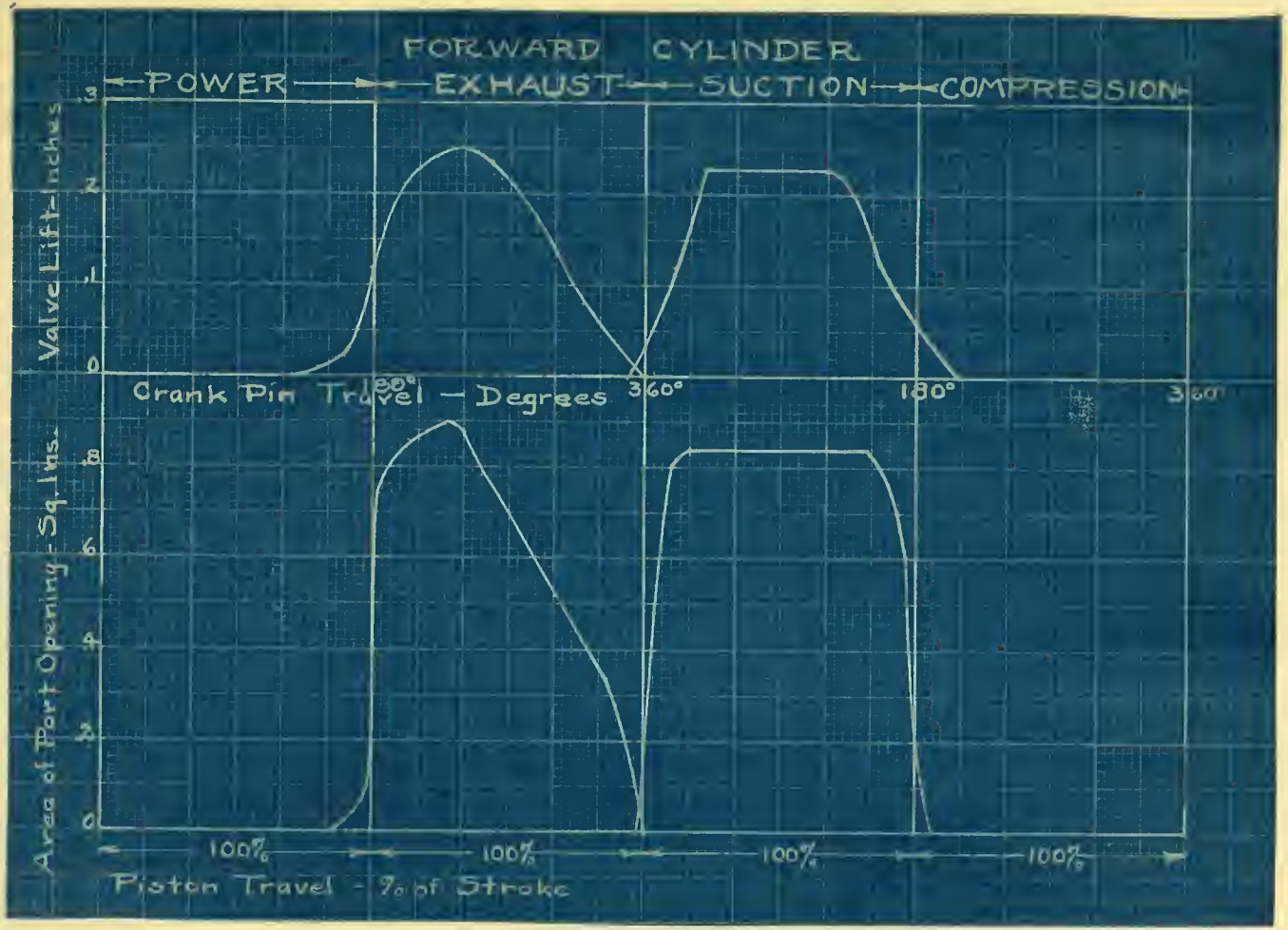

FORWARD CYLINDER
POWER
EXHAUST
SUCTION
COMPRESSION
Valve Lift-Inches
3
2
1
0
Crank Pin Travel — Degrees
180°
360°
180°
360°
Area of Port Opening - Sq. Ins.
8
6
4
2
0
100%
100%
100%
100%
Piston Travel - % of Stroke

FIGURE 19.

TABLE 9.

VALVE LIFT DATA FOR HARLEY-DAVIDSON TWIN.

BACK CYLINDER, INLET VALVE.

CRANK PIN DEG. EAST D.C.	LIFT INS.	PORT OPEN-ING-INS.	% OF PISTON TRAVEL
- 8°	0 "	0	.1
+ 15	.075	.279	4.5
35	.135	.512	13.0
55	.18	.675	28
75	.20	.741	49
95	.21	.78	63
115	"	"	78
135	"	"	91
155	"	"	95
175	.155	.512	98
195	.02	.071	99
215	0	0	12

BACK CYLINDER, EXHAUST VALVE.

156	0	0	90
155	.03	.126	96.5
175	.10	.369	99
195	.17	.592	5
215	.22	.784	13
255	.27	.974	27
255	.29	1.052	35
275	.29	1.052	49
295	.26	.912	65
315	.21	.747	82
335	.10	.369	94
3	0	0	1

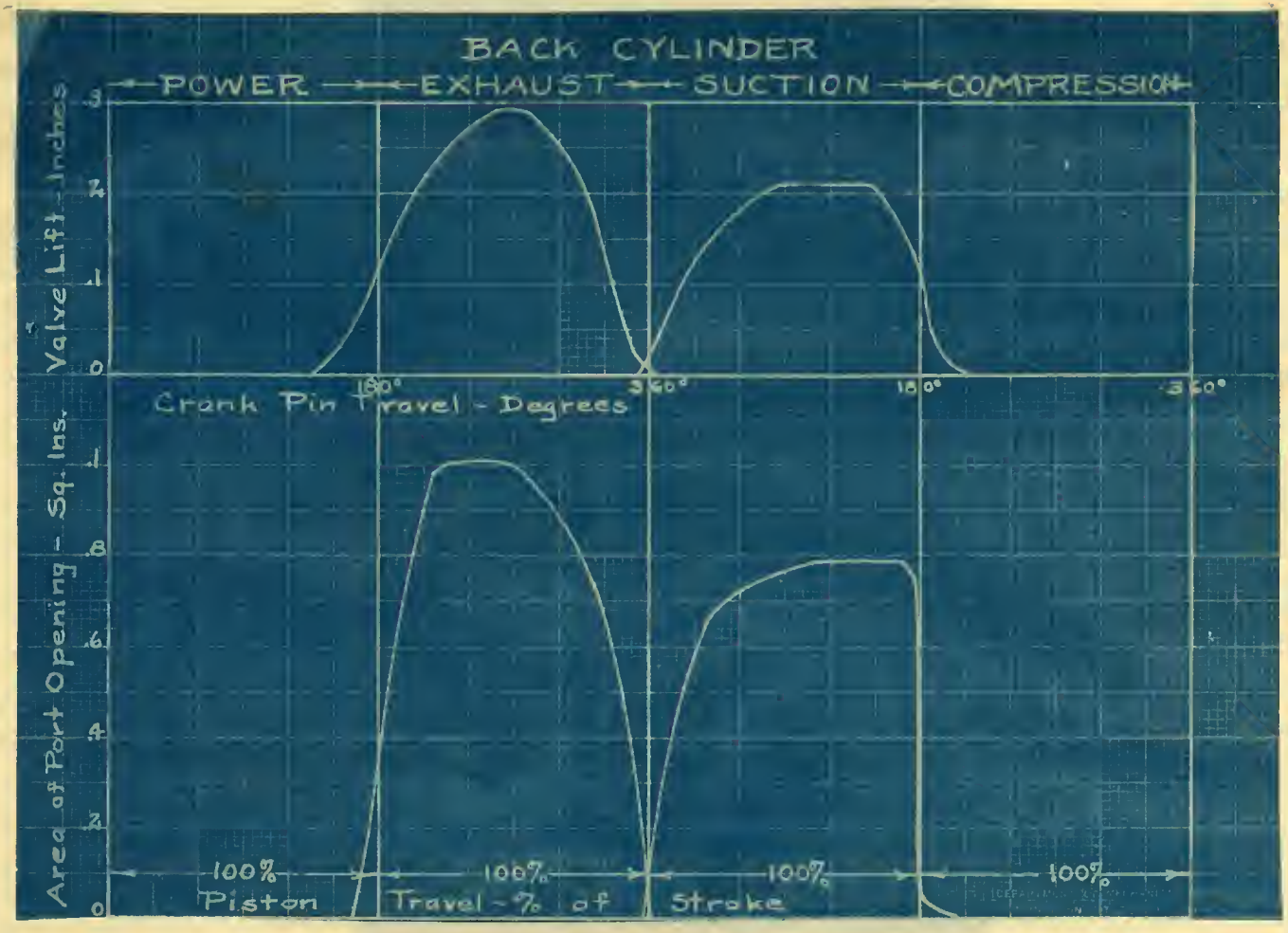

BACK CYLINDER
POWER
EXHAUST
SUCTION
COMPRESSION
Valve Lift - Inches
3
2
1
0
180°
360°
180°
360°
Crank Pin Travel - Degrees
Area of Port Opening - Sq. Ins.
1
.8
.6
.4
.2
0
100% Piston
100% Travel - % of
100% Stroke
100%

FIGURE 20.

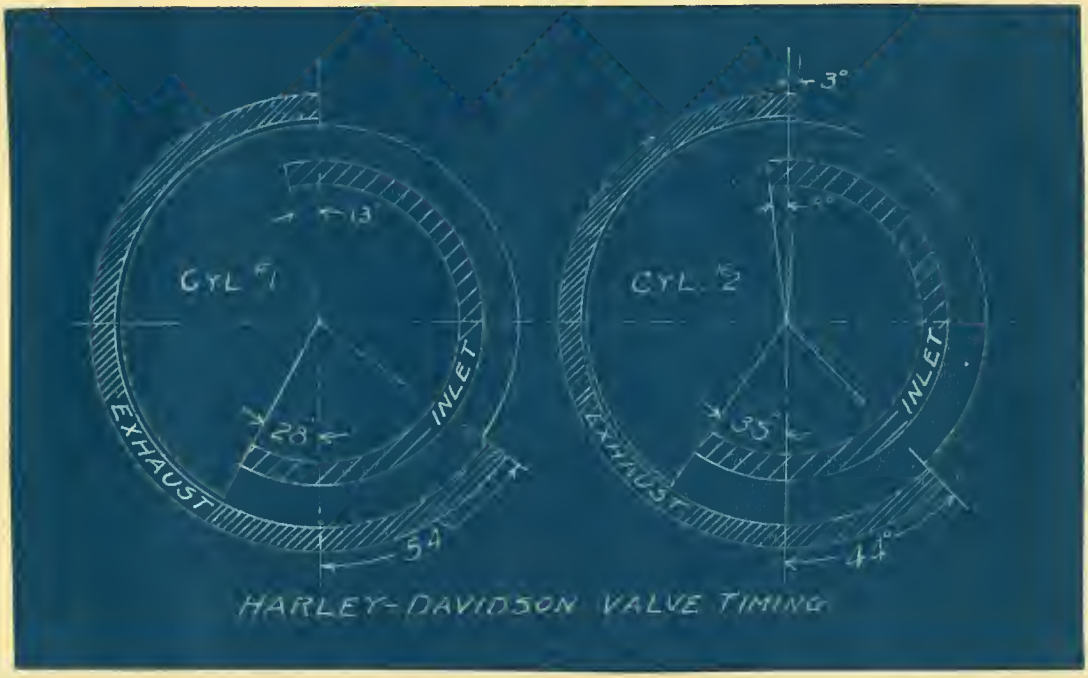

3°
13°
CYL. 1
CYL. 2
INLET
EXHAUST
28°
35°
54°
44°
HARLEY-DAVIDSON VALVE TIMING

FIGURE 21.

CHAPTER 4.

DISCUSSION OF RESULTS.

Perhaps the first thing which struck the observers of these two tests was the necessity of having an engine stand of sufficient mass to damp out excessive vibration. In testing the Thor engine, this vibration was never cut down enough to give the engine a fair chance. At first the test was made on a heavy cast iron table (Fig. 5), but even after removing the engine to a cement floor and clamping it to imbedded iron slots (Fig. 3) the vibration was still annoying. At high speeds the gas seemed to hold back a little, and this was probably due to the gasoline spurting thru the carburetor nozzle instead of flowing in a steady stream.

The performance of the Thor is shown clearly in Figures No. 12 and 13. The brake horse power keeps above the S.A.E. rating un-

til 2100 r.p.m. The friction horse power, owing to the small stock clearance, is rather high, going up in a steep curve. The mean effective pressure net is very good at low speeds, being 74 pounds per square inch at 1600 r.p.m., but at higher speeds it drops off quite rapidly, being only 45 pounds per sq.in. at 2800 r.p.m.

The inlet valve opening is one sq. in. per 92.5 cubic inches displacement. The new Packard 300 cu.in. aviation motor, which runs at about the same speed as these motorcycle engines, has one sq.in. inlet valve opening for every 17 cu.ins. displacement; and also develops about one horse power for every two cubic inches displacement. The Thor Twin tested developed one horse power for every 5.4 cu.ins. displacement.

Performance curves for the Harley-Davidson are shown in Figures 17 and 18. The brake horse power curve keeps above the S.A.E.

rating until 2550 r.p.m. The mean effective pressure is high, being 75.5 pounds per sq. in. at 1400 r.p.m. and still 64 pounds per sq .in. at 2800 r.p.m. The inlet valve opening is one square inch per 43 cu.ins. displacement, and this accounts considerably for the high power of this engine. The friction horse power is low, owing somewhat to the racing clearance, and it increases in a straight line ratio with the speed. One horse power was developed for each 4.32 cubic inches displacement.